Available online at www.sciencedirect.com
ScienceDirect
Renewable and Sustainable Energy Reviews
12 (2008) 1383–1401
RENEWABLE & SUSTAINABLE ENERGY REVIEWS
www.elsevier.com/locate/rser

Exergetic assessment of direct-expansion solar-assisted heat pump systems: Review and modeling

Ozer Kara[a,*], Koray Ulgen[a], Arif Hepbasli[b]

[a]*Solar Energy Institute, Ege University, 35100 Izmir, Turkey*
[b]*Department of Mechanical Engineering, Ege University, 35100 Izmir, Turkey*

Received 23 November 2006; accepted 15 December 2006

Abstract

Although the idea of using a solar collector as the evaporator in the traditional heat pump cycle was first proposed in the year 1955, studies on the subject began in the late 70s. One of the keystones for obtaining sustainable development is also the use of exergy analysis. In this regard, the main objectives in doing the present study are twofold, namely (i) to review studies on direct-expansion solar-assisted heat pump systems (DX-SAHPs) and (ii) to present a mathematical model along with an illustrative example, which is used for heating an office space in Solar Energy Institute of Ege University, Izmir, Turkey, by floor heating with a DX-SAHP system. The system uses a $4\,m^2$ bare flat-plate collector as the evaporator, while the working fluid is chosen to be R-22. Water is heated by the heat pump and heat is delivered to the office space by floor heating. Exergy equations for the system are derived, while exergy calculations are made. The exergy efficiency values for the individual components of the DX-SAHP system are found to range from 10.74% to 88.87%. It is expected that this study will be very beneficial to everyone involved or interested in the exergetic design, analysis and performance assessment of DX-SAHPs.

Keywords: Exergy analysis; Direct expansion; Solar-assisted heat pump; Sustainability

*Corresponding author. Tel.: +90 232 388 6023; fax: +90 232 388 6027.
E-mail address: ozer.kara@ege.edu.tr (O. Kara).

1364-0321/$ - see front matter
doi:10.1016/j.rser.2006.12.001

Contents

1. Introduction

Energy consumption plays the major role in the overall budget in many countries. Domestic consumption of energy mainly occurs in hot water production and space heating. Sources for energy used generally in these applications are mainly fossil fuels. Using renewable energy sources, mainly solar energy, for especially domestic applications may be the key to reduce energy costs and consequently the amount of energy costs within the whole country budget, while providing sustainable development.

Heat pump systems are heat-generating devices that can be used to heat water to be used in either domestic hot water or space heating applications. For heat pumps, a basic factor of great importance for its successful application is the availability of a cheap, dependable heat source for the evaporator—preferably one at relatively high temperature. The coefficients of performance (COP) of heat pump systems depend on many factors, such as the temperature of low-energy source, the temperature of delivered useful heat, the working medium used, the characteristics of components of heat pump systems, etc. Among the above mentioned, the temperature of the evaporator is a key factor [1].

In order to improve the heat pump COP_H for heating and displace the fossil energy resource, the idea of combining the heat pump and solar energy in mutual beneficial ways has been proposed and developed by several researchers [2–5]. In a typical solar-assisted heat pump (SAHP), the closed loop of the solar collector mainly uses water or air as the working fluid and is separated from the heat pump evaporator. In a direct-expansion SAHP (DX-SAHP), the collector and evaporator functions are combined into one unit, where the refrigerant from the condenser gets evaporated by incident solar energy [6].

The main objectives in doing the present study are to: (i) review studies that have been conducted on DX-SAHP systems taking into consideration the analysis that have been made (theoretical–experimental, energy–exergy) and the utilization of the systems (space heating, water heating, air conditioning (A/C)), (ii) present a mathematical model for exergy-based DX-SAHP calculations including thermodynamic parameters such as fuel depletion ratio, relative irreversibility, productivity lack and exergetic factor as well as improvement potential and (iii) apply the presented mathematical model to an illustrative example.

Nomenclature

COP	coefficient of performance (dimensionless)
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
f	exergetic factor (dimensionless)
$\dot{F}$	exergetic fuel rate (kW)
h	specific enthalpy (kJ/kg)
$\dot{I}$	irreversibility (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
$\dot{P}$	exergetic product rate (kW)
$\dot{Q}$	heat transfer rate (kW)
s	specific entropy (kJ/kg K)
T	temperature (K or °C)
$\dot{W}$	work rate or power (kW)

Greek letters

ψ	specific exergy (kJ/kg)
η	energy (first law) efficiency (dimensionless)
δ	fuel depletion rate (dimensionless)
ε	exergy (second law) efficiency (dimensionless)
ξ	productivity lack (dimensionless)
χ	relative irreversibility (dimensionless)

Indices

comp	compressor
cond	condenser
coll	collector
dest	destroyed (destruction)
elec	electric
evap	evaporator
H	heating
in	inlet
int	internal
mech	mechanical
out	outlet
pump	circulating pump
r	refrigerant
s	solar radiation
T	total
u	useful
w	water

0	dead (reference) state
·	rate

Abbreviations

A/C	air conditioning
DX-SAHP	direct-expansion solar-assisted heat pump
SAHP	solar-assisted heat pump
TEV	thermostatic expansion valve

This paper is organized as follows. Section 1 is a brief introduction to the study, while the studies conducted on the subject is reviewed in Section 2. A mathematical model is given and related results obtained from calculations are presented in Section 3, while the last section concludes.

2. Reviewing studies conducted on DX-SAHP systems

Solar energy can be used to evaporate the refrigerant in a heat pump cycle by using a solar collector as the evaporator [7]. This, by increasing the evaporation temperature of the refrigerant, leads to increased heat pump performance. The heat pump cycle, in which a solar collector is used as the evaporator, is then called a DX-SAHP system.

2.1. Historical development

The oil crisis in the beginning of the 70s has led researchers to use alternative energy sources for energy production. Solar energy was the first choice. Later, when heat pumps became popular for heating and cooling applications, combining solar energy and heat pumps in several ways were proposed by many researchers. Although the idea of a DX-SAHP was first proposed by Sporn and Ambrose in 1955, because the first main studies on the subject began at that date, the beginning of the studies on DX-SAHP systems is therefore accepted as the late 70s.

Charter and Taylor [8] proposed a DX-SAHP system with an unglazed solar collector as the evaporator. Following this study, several types of collectors were proposed to be used in DX-SAHP systems and studies were developed to find out their applicability [2–5].

O'Dell et al. [9] expressed the heat gain of the condenser and the COP as a function of evaporation temperature. Using these expressions along with the collector heat gain, condenser heat gain and COP were calculated under different ambient conditions.

Chaturvedi and Shen [10] investigated the heat performance of a DX-SAHP system. An unglazed collector was used as the evaporator and R12 was used as the refrigerant. COP and collector efficiency at different ambient and working conditions were found to be 2.0–3.2 and 40–70%, respectively.

Long-term heat performance analysis of a DX-SAHP system was performed by Chaturvedi and Abazeri [11]. Simulative calculations showed that the heat performance of

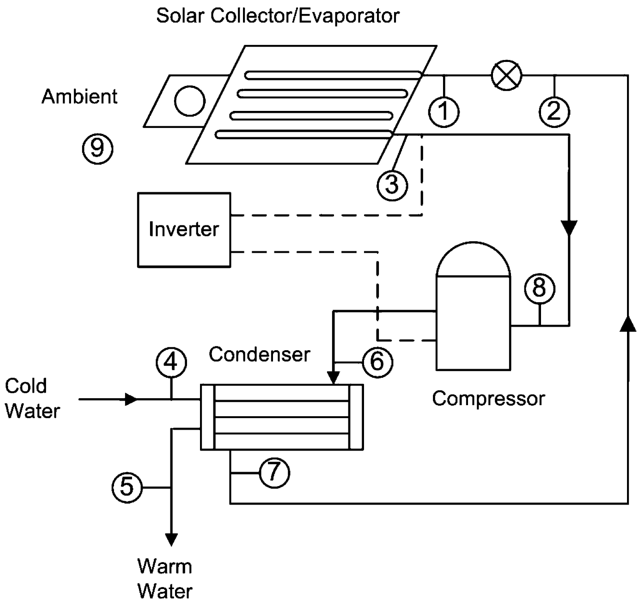


Fig. 1. A DX-SAHP system used for domestic hot water production drawn by Chaturvedi et al. [12].

the system was mainly effected by collector area, compressor period, target temperature and thermophysical properties of the refrigerant used; the other parameters had no significant effect on the long-term system performance.

Chaturvedi et al. [12] used a bare flat-plate solar collector as the evaporator for a heat pump that is used for domestic hot water application, as shown in Fig. 1. Experimental results showed that the COP of the system could be improved significantly by lowering the compressor speed due to ambient conditions.

Torres-Reyes et al. [13] presented a theoretical and experimental exergy analysis of a SAHP, which was used for air heating. They proposed a methodology to optimize the evaporation and condensing temperatures of the working fluid (R-22).

Ito et al. [7] also used an unglazed solar collector as the evaporator. Although there were two types of solar collectors in the experimental setup (Fig. 2), namely convective type and radiative type (flat plate) (Fig. 3), only the data for the radiative type were examined. Electricity consumption of the compressor and COP were defined as functions of evaporation temperature and condenser inlet temperature of the refrigerant. Evaporation temperature was found to be 17 °C above the ambient and COP was found to be 5.3, while condenser inlet temperature of the refrigerant was 40 °C.

Torres-Reyes and Cervantes de Gortari [14] investigated the thermal performance of a DX-SAHP system (Fig. 4). Exergetic efficiencies were found to be in the range of 0.067–0.13 and 0.083–0.14, which were calculated for two different equations, while the COP values for the heat pump varied from 2.56 to 4.36.

Hawlader et al. [15] also performed experimental and analytical studies on a DX-SAHP system, which used a bare flat-plate solar collector as the evaporator and R-134a as the working fluid. The COP values were found to range from about 4 to 9, while the solar collector efficiency varied between 40% and 75%.

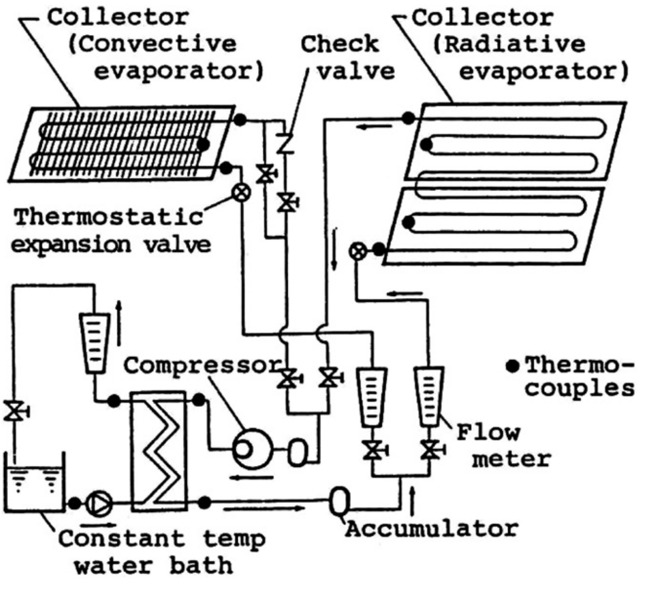


Fig. 2. DX-SAHP setup with two different types of solar collectors used as evaporator drawn by Ito et al. [7].

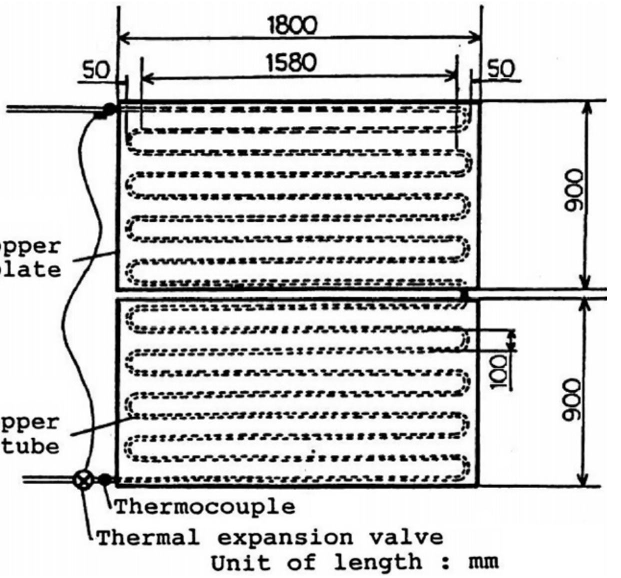
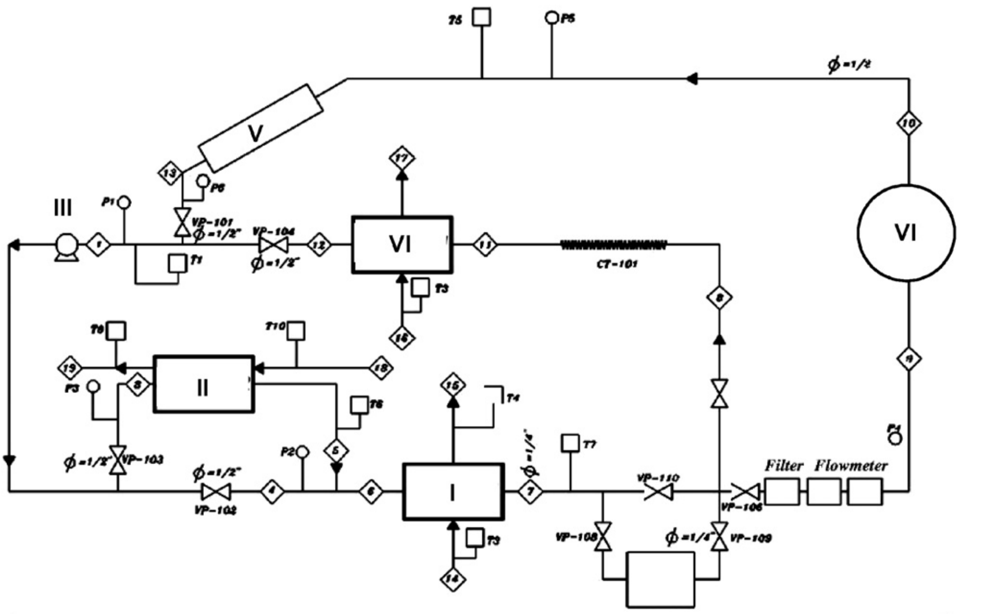


Fig. 3. Radiative type (flat-plate) solar collector drawn by Ito et al. [7].

Cervantes de Gortari and Torres-Reyes [16] conducted experiments on a DX-SAHP system and determined the maximum exergy efficiency. The results showed that the most irreversibility occurred in the evaporator of the heat pump, which was the solar collector.

Chyng et al. [17] investigated the performance of an integral-type DX-SAHP system (Fig. 5), in which the heat pump, solar collector and the water storage tank were designed



I: condenser, II: condenser, III: hermetic compressor, IV: thermal expansion valve, V: solar flat collector, VI: conventional cross-flow heat exchanger

Fig. 4. Experimental setup for a heat pump used as both as a conventional heat pump or a DX-SAHP drawn by Torres-Reyes and Cervantes de Gortari [14].

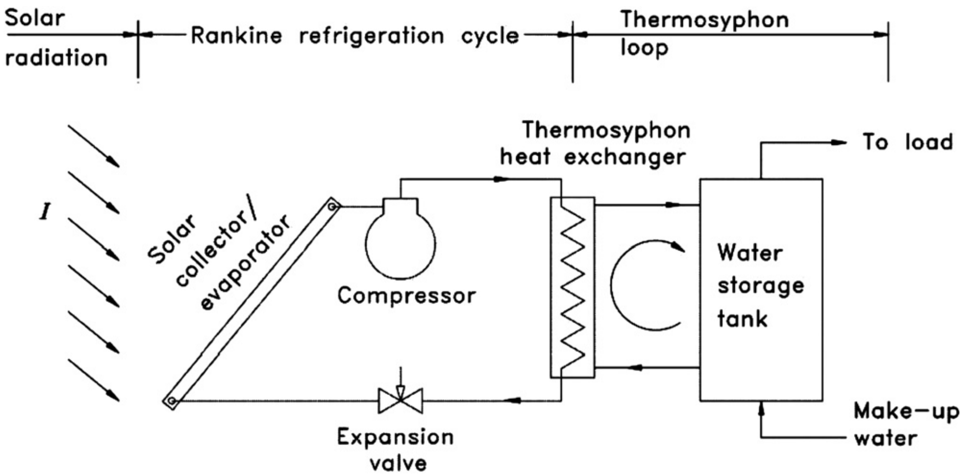


Fig. 5. Schematic diagram of the integral-type DX-SAHP system drawn by Chyng et al. [17].

to form a single unit that was considered to be very easy to install (Fig. 6). The COP of the system was found to be around 1.7–2.5 throughout the year depending on ambient conditions.

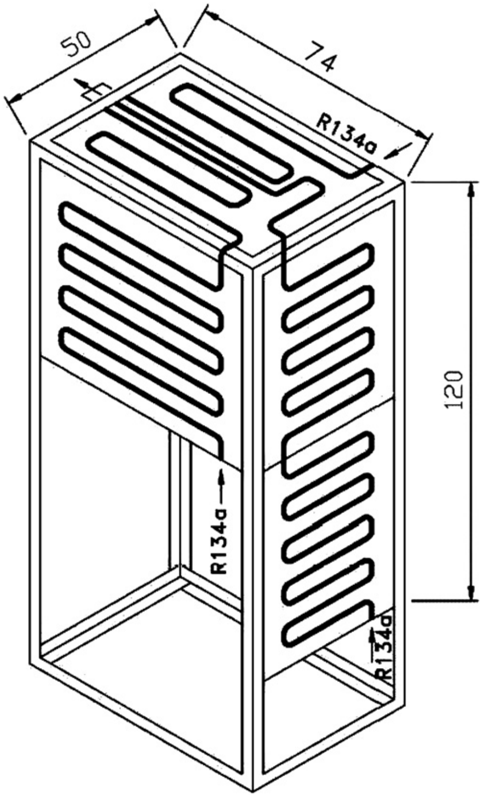


Fig. 6. The heat pump, solar collector and the water storage tank combined to form a single unit drawn by Chyng et al. [17]. All dimensions are in cm.

Kuang et al. [6] used an unglazed solar collector of $2\,m^2$ as the evaporator (Fig. 7). To estimate long-term system performance, simulative calculations were carried out. COP and collector performance were found to be 4–6 and 40–60%, respectively. Experimental analysis results for winter periods were in a good agreement with the results from the simulation process.

Gorozabel Chata et al. [18] analyzed the thermal performance of a DX-SAHP system with two different solar collector configurations using various refrigerants, including pure refrigerants and refrigerant mixtures, namely R-134a, R-404a, R-407c and R-410a. The results were then compared with the results from R-12 and R-22 to achieve a comparative analysis. The results showed that R-12 produced the highest value of COP, followed by R-22 and R-134a.

Ito et al. [19] studied on a DX-SAHP system involving a PV/T solar collector of $1.9\,m^2$ as the evaporator. The collector efficiency factor was obtained to be 0.9 for the evaporator.

Kuang and Wang [20] reported on the long-term performance of a DX-SAHP system with a bare flat-plate solar collector as the evaporator, which could be used for space heating in winter, air-conditioning in summer and hot water production during the whole

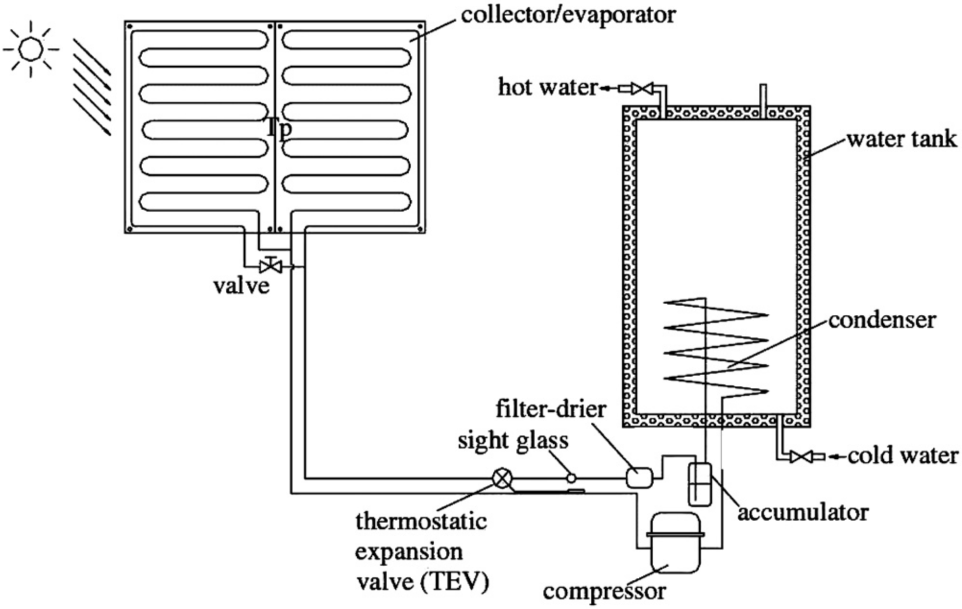


Fig. 7. Schematic of the DX-SAHP used for water heating drawn by Kuang et al. [6].

year. The COP values for the system ranged from 2.1 to 2.7 for space-heating-only mode throughout the year, while it could produce 200–1000 l of hot water daily at a temperature of 50 °C under various weather conditions.

2.2. Studies conducted

A detailed review of studies conducted on DX-SAHP systems is given in Table 1. As can be seen from the table, mainly bare flat-plate collectors are used by many investigators, while very few of them have been conducted with various types of solar collectors, such as glass-covered and PV/T collectors. This is probably because a bare flat-plate solar collector generally generates enough heat to be used as the evaporator in DX-SAHP systems, reducing the need for other types of solar collectors, such as glass-covered, selective-surface solar collectors, which would increase the total installation costs of the systems.

Collector area used in the studies ranged from 1.82 to 10.5 m^2, providing a relatively wide range, while studies dealing with collector efficiencies were a few in numbers.

Refrigerants used were mainly R-12 and R-22, while the performances of systems with refrigerants, such as R-134a, R-404a, R-407c and R-410a were rarely investigated. In case R-12 and R-22 are either banned or non-produced refrigerants due to environmental aspects, the number of studies involving authorized refrigerants should be increased.

The performance indicator for heat pump systems, namely COP values, varied from 1.5 to 9, while most of the values were between 2 and 3. These are very close values, as expected, to traditional heat pump systems available in the market.

Most of the systems were used in water-heating applications. Space heating and A/C performances were investigated by very few researchers. Since energy is mainly used for water and space-heating purposes in many countries, studies related with improving the

Table 1
Studies on DX-SAHP systems

#	Year	Investigator	Location	Collector type	Collector area (m^2)	Type of refrigerant	COP[a]	Collector efficiency (%)	Type of study		Type of analysis		Type of system		
									Theoretical (simulation)	Experimental	Energy	Exergy	Space heating	Water heating	A/C[c]
1	1984	Chaturvedi and Shen [10]	Norfolk	Bare flat plate, Black painted	3.39	R-12	2–3	40–70	√	√	√	X		√	
2	1987	Chaturvedi and Abazeri [11]	Norfolk	Bare flat plate	2	R-12, R-22	1.5–5	N/A[b]	√	X	√	X		√	
3	1998	Chaturvedi et al. [12]	Norfolk	Bare flat plate	3.48	R-12	2.5–4.0	N/A	√	√	√	√		√	
4	1998	Torres-Reyes et al. [13]	Guanajuato, Mexico	Bare flat plate	4.5	R-22	N/A	N/A	√	√	X	√		√	
5	1999	Hulin, et al. [1]	Kagoshima, Japan	Bare flat plate	?		3–3.5	N/A	X	√	√	X		√	
				Glass covered	1.82		3–8.25	N/A							
6	1999	Ito et al. [7]	Japan	Bare flat-plate	3.24	R-12	5.3	N/A	√	√	√	X		√	
7	2001	Torres-Reyes and Cervantes de Gortari [14]	Guanajuato, Mexico	Flat-plate	4.5	R-22	2.56–4.36	N/A	√	√	X	√		√	
8	2001	Hawlader et al. [15]	Singapore	Bare flat plate	3 (1.5 × 2)	R-134a	4–9	40–75	√	√	√	X		√	
9	2002	Cervantes de Gortari and Torres-Reyes [16]	Guanajuato, Mexico	Bare flat-plate	4.5	R-22	N/A	N/A	√	√	X	√		√	
10	2003	Chyng et al. [17]	Taiwan	Bare flat plate	1.86	R-134a	1.7–2.5	N/A	√	√	√	X		√	
11	2003	Kuang et al. [6]	Shanghai, China	Bare flat plate, Selective Surface	2	R-22	4–6	40–60	√	√	√	X		√	
12	2005	Gorozabel Chata et al. [18]		Bare flat plate	N/A	R-12, R-22, R-134a, R-404a, R-407c, R-410a	2.5–5	N/A	√	X	√	X		√	
				One cover plate			N/A	N/A							
13	2005	Ito et al. [19]	Japan	PV/T	1.9	R-22	4.5–6.5	N/A	√	√	√	X	√	√	
14	2006	Kuang and Wang [20]	Shanghai, China	Bare flat-plate	10.5	R-22	2.6–3.3 (HP), 2.1–2.7 (sys.)	N/A	X	√	X	X	√	√	√

[a]COP: Coefficient of performance.
[b]N/A: Not available.
[c]A/C: Air conditioning.

performances and reducing costs in such applications play an important role, also providing less demand for fossil fuels. Theoretical and experimental analysis as well as energy analysis were conducted by nearly all of the studies, while exergy analysis was covered by very few of them.

3. An illustrative example

In this subsection, a DX-SAHP system is presented as an illustrative example for modeling and performance evaluation purposes.

3.1. System description

The DX-SAHP system basically consists of a solar collector, a heat exchanger as condenser, a thermostatic expansion valve and a compressor. The solar collector is used as the evaporator of the heat pump system. The refrigerant is directly vaporized in the solar collector–evaporator due to the solar energy input, where phase change from liquid to vapor occurs. Thus, unlike the conventional SAHP systems, where two separate system components are used for the same purpose, both processes, namely collecting solar energy and vaporizing the refrigerant, are realized in one unit only. This leads to several advantages compared to conventional SAHP systems.

(a) The direct vaporization of the refrigerant in the solar collector–evaporator leads to higher heat transfer coefficients.
(b) The use of the solar collector as the evaporator reduces overall system cost because the need for an additional evaporator in the traditional SAHP system is eliminated.
(c) Problems, which may occur in water collectors (i.e. corrosion, night freezing), are eliminated due to the use of refrigerants as the working fluid, leading to longer system life.
(d) Using refrigerants as the working fluid in the heat pump cycle results with low temperature during the evaporation process in the solar collector, which leads to lower system losses since the collector loss value is a function of the collector to ambient temperature difference.
(e) The collector, including bare flat-plate collectors, works at high efficiency values based on the low collector to ambient temperature differences, which also reduces collector cost.

A typical DX-SAHP system is illustrated in Fig. 8. The system works based on the vapor compression cycle. The refrigerant is vaporized while passing through the solar collector (IV), which may be represented as the $4\rightarrow1$ process. The vaporized refrigerant is then compressed in the compressor (I) ($1\rightarrow2$ process) and sent to the condenser (II) in order to be condensed by rejecting the useful heat content to another heat transfer medium (i.e. air, water) ($2\rightarrow3$). After being condensed, the high-pressure refrigerant passes through a thermostatic expansion valve (III), where it undergoes the throttling process ($3\rightarrow4$) being ready to enter the collector as low-pressure liquid and thus finishing the cycle.

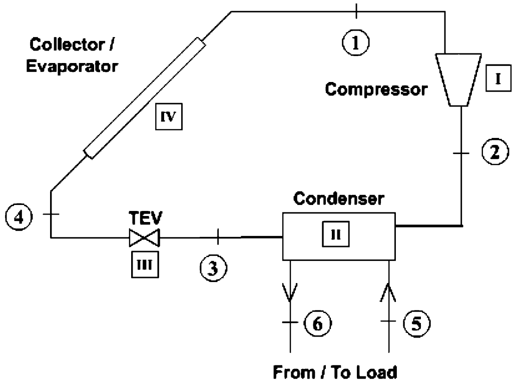


Fig. 8. Direct-expansion solar-assisted heat pump (DX-SAHP).

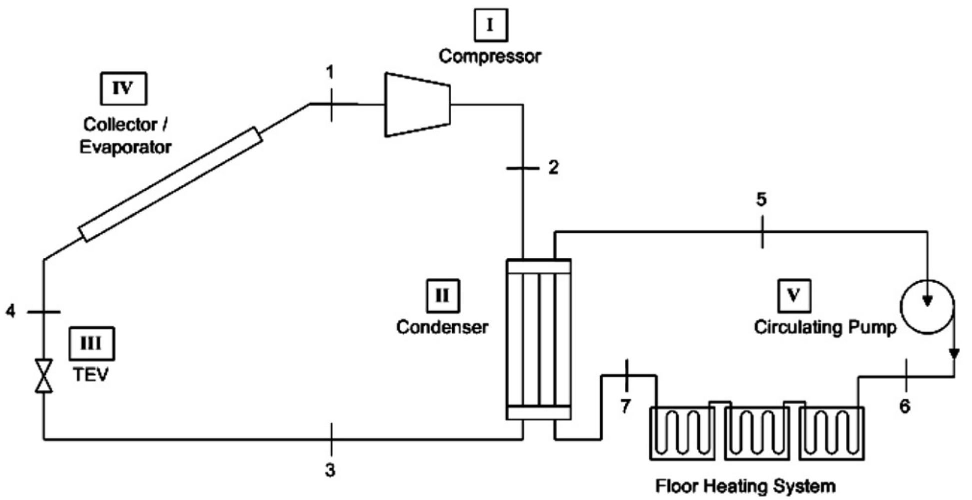


Fig. 9. Schematic of the DX-SAHP floor heating system.

3.2. Modeling

Energy and exergy analysis is presented for the performance evaluation of the DX-SAHP systems. Considering the general balance equations as well as the relations for thermodynamic parameters, exergetic improvement potential rate and general energy and exergy efficiencies have been given by Ozgener and Hepbasli [21].

The exergy rate is given by

$$\dot{E}x = \dot{m}\psi \tag{1a}$$

with the flow exergy or specific exergy

$$\psi = (h - h_0) - T_0(s - s_0), \tag{1b}$$

where h is the enthalpy, s is the entropy and the subscript zero indicates properties at the restricted dead state of P_0 and T_0.

Mass and energy balances as well as exergy destructions obtained from exergy balances for each of the heat pump components illustrated in Fig. 9 are derived as follows:

Compressor (I):

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{\mathrm{r}}, \tag{2a}$$

$$\dot{W}_{\mathrm{comp}} = \dot{m}_{\mathrm{r}}(h_2 - h_1), \tag{2b}$$

$$\dot{W}_{\mathrm{comp,elec}} = \frac{\dot{W}_{\mathrm{comp}}}{\eta_{\mathrm{comp,mech}}\eta_{\mathrm{comp,elec}}}, \tag{2c}$$

$$\dot{E}x_{\mathrm{dest,comp}} = \dot{m}_{\mathrm{r}}(\psi_1 - \psi_2) + \dot{W}_{\mathrm{comp,elec}}, \tag{2d}$$

where heat interactions with the environment are neglected.

The mechanical–electrical losses:

$$\dot{E}x_{\mathrm{dest,comp,mech,elec}} = \dot{W}_{\mathrm{comp,elec}}(1 - \eta_{\mathrm{comp,elec}}\eta_{\mathrm{comp,mech}}). \tag{2e}$$

The internal reversibility due to fluid friction:

$$\dot{E}x_{\mathrm{dest,comp,int}} = \dot{E}x_{\mathrm{dest,comp}} - \dot{E}x_{\mathrm{dest,comp,mech,elec}}, \tag{2f}$$

where $\eta_{\mathrm{comp,mech}}$ and $\eta_{\mathrm{comp,elec}}$ are the compressor mechanical and the compressor motor electrical efficiencies, respectively [22].

$$\varepsilon_{\mathrm{comp}} = \frac{\dot{E}x_2 - \dot{E}x_1}{\dot{W}_{\mathrm{comp,elec}}} = \frac{\dot{m}_{\mathrm{r}}(\psi_2 - \psi_1)}{\dot{W}_{\mathrm{comp,elec}}}. \tag{2g}$$

Condenser (II):

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{\mathrm{r}}, \tag{3a}$$

$$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{r}}(h_2 - h_3), \tag{3b}$$

$$\dot{E}x_{\mathrm{dest,cond}} = \dot{m}_{\mathrm{r}}(\psi_2 - \psi_3) + \dot{m}_w(\psi_6 - \psi_5) \tag{3c}$$

with $\dot{E}x_{\mathrm{heat,cond}} = 0$.

$$\varepsilon_{\mathrm{cond}} = \frac{\dot{E}x_3}{\dot{E}x_2} = \frac{\psi_3}{\psi_2}. \tag{3d}$$

Thermostatic expansion valve (TEV) (III):

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{\mathrm{r}}, \tag{4a}$$

$$(h_3 = h_4), \tag{4b}$$

$$\dot{E}x_{\mathrm{dest,TEV}} = \dot{m}_{\mathrm{r}}(\psi_3 - \psi_4), \tag{4c}$$

$$\varepsilon_{\mathrm{TEV}} = \frac{\dot{E}x_4}{\dot{E}x_3} = \frac{\psi_4}{\psi_3}. \tag{4d}$$

Solar collector (IV):

The instantaneous exergy efficiency of the solar collector can be defined as the ratio of the increased refrigerant exergy to the exergy of the solar radiation [23]. In other words, it is a ratio of the useful exergy delivered to the exergy absorbed by the solar collector [24].

$$\varepsilon_{\mathrm{coll}} = \frac{\dot{E}x_{\mathrm{u}}}{\dot{E}x_{\mathrm{coll}}} \tag{5a}$$

with

$$\dot{E}x_{\mathrm{u}} = \left[\dot{m}_{\mathrm{r}}\left(h_{\mathrm{r,out}} - h_{\mathrm{r,in}}\right)\left(\frac{T_0 - T_{\mathrm{evap}}}{T_{\mathrm{evap}}}\right)\right] \tag{5b}$$

and

$$\dot{E}x_{\mathrm{coll}} = AI_{\mathrm{T}}\left[1 + \frac{1}{3}\left(\frac{T_0}{T_{\mathrm{s}}}\right)^4 - \frac{4}{3}\left(\frac{T_0}{T_{\mathrm{s}}}\right)\right], \tag{5c}$$

where T_{s} is the solar radiation temperature and taken to be 6000 K, while the Petela expression is used in calculating the exergy of solar radiation as the exergy input to the solar collector [25].

Circulating pump (V):

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_{\mathrm{w}}, \tag{6a}$$

$$\dot{W}_{\mathrm{pump}} = \dot{m}_{\mathrm{w}}(h_6 - h_5), \tag{6b}$$

$$\dot{W}_{\mathrm{pump,elec}} = \frac{\dot{W}_{\mathrm{pump}}}{(\eta_{\mathrm{pump,elec}}\eta_{\mathrm{pump,mech}})}, \tag{6c}$$

$$E\dot{x}_{\mathrm{dest,pump}} = \dot{m}_{\mathrm{w}}(\psi_5 - \psi_6) + \dot{W}_{\mathrm{pump,elec}}, \tag{6d}$$

where heat interactions with the environment are neglected.

The mechanical–electrical losses:

$$\dot{E}x_{\mathrm{dest,pump,mech,elec}} = \dot{W}_{\mathrm{pump,elec}}(1 - \eta_{\mathrm{pump,elec}}\eta_{\mathrm{pump,mech}}). \tag{6e}$$

The internal reversibility due to fluid friction:

$$\dot{E}x_{\mathrm{dest,pump,int}} = \dot{E}x_{\mathrm{dest,pump}} - \dot{E}x_{\mathrm{dest,pump,mech,elec}}, \tag{6f}$$

where $\eta_{\mathrm{pump,mech}}$ and $\eta_{\mathrm{pump,elec}}$ are the compressor mechanical and the compressor motor electrical efficiencies, respectively [22].

$$\varepsilon_{\mathrm{pump}} = \frac{\dot{E}x_6 - \dot{E}x_5}{\dot{W}_{\mathrm{pump,elec}}} = \frac{\dot{m}_w(\psi_6 - \psi_5)}{\dot{W}_{\mathrm{pump,elec}}}. \tag{6g}$$

Table 2
Exergetic data provided for the DX-SAHP system

State no.	Name of element	Fluid	Temperature T (°C)	Pressure P (kPa)	Specific enthalpy h (kJ/kg)	Specific entropy s (kJ/kg K)	Mass flow rate $\dot{m}$ (kg/s)	Specific exergy ψ (kJ/kg)	Exergy rate $\dot{E}x$ (kW)	Energy rate $\dot{E}$ (kW)
0	Dead state (R-22)	R-22	2	101.33	414.62	1.9309	—	0	—	—
0′	Dead state (water)	R-22	2	101.33	8.1	0.0296	—	0	—	—
1	Compressor inlet	R-22	−5	421.1	403.51	1.7594	0.036	36.0443	1.2976	14.5264
2	Compressor outlet/condenser inlet	R-22	101.3	2,173.20	463.27	1.8086	0.036	82.2743	2.9619	16.6777
3	Condenser outlet/ TEV inlet	R-22	55	2,173.20	270.31	1.2291	0.036	48.6768	1.7524	9.7312
4	TEV inlet/ evaporator inlet	R-22	−5	421.1	270.31	1.2488	0.036	43.2593	1.5573	9.7312
5	Circulating pump inlet/condenser outlet	Water	50	250	209.69	0.7046	0.165	15.9650	2.6342	34.5989
6	Floor heating inlet/circulating pump outlet	Water	50.20	250	210.53	0.7072	0.165	16.0900	2.6549	34.7375
7	Floor heating outlet condenser inlet	Water	40	250	167.46	0.5719	0.165	10.2275	1.6875	27.6309

3.3. Results and discussion

The above modeled DX-SAHP system is considered to be used as the heat resource of a floor heating system in Solar Energy Institute of Ege University in Izmir, Turkey. The schematic diagram of the system is shown in Fig. 9.

Exergy calculations were performed according to the model presented in the previous subsection, while the following assumptions were used during the calculations:

(a) All processes are steady state and steady flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.
(b) Heat transfer to the system and work transfer from the system are positive.
(c) Heat transfer and refrigerant pressure drops in the tubing connecting the components are neglected since their lengths are short.
(d) The compressor mechanical ($\eta_{\text{comp,mech}}$) and the compressor motor electrical ($\eta_{\text{comp,elec}}$) efficiencies are 90% and 84%, respectively.
(e) The circulating pump mechanical ($\eta_{\text{comp,mech}}$) and electrical ($\eta_{\text{comp,elec}}$) efficiencies are 82% and 88%, respectively.
(f) Evaporation and condensing temperatures are -5 and 55 °C, respectively, while no superheating and subcooling occurs.
(g) Solar collector calculations were made according to weather data measured in Solar Energy Institute of Ege University in Izmir, Turkey. Based on the data measured, solar radiation is taken to be 140 W/m^2 at the day when ambient temperature is 2 °C, which is selected to be equal to the reference state temperature.
(h) Solar collector area used in the system is taken to be 4 m^2.
(i) The reference state temperature and pressure are taken to be 2 °C and 101.325 kPa, respectively.

The exergetic data used and results of the exergy calculations made are listed in Tables 2 and 3, respectively.

The greatest irreversibility (exergy destruction) occurs in the compressor, followed by the condenser, solar collector, TEV and circulating pumps, as seen in Table 3.

The mechanical–electrical losses in the compressor are found to account for 24.2% of the system input. The mechanical–electrical losses are due to imperfect electrical, mechanical and isentropic efficiencies and emphasize the need for paying close attention to the selection of this equipment, since components of inferior performance can considerably reduce overall system performance. Since compressor power depends strongly on the inlet and outlet pressures, any heat exchanger improvements that reduce the temperature difference will reduce compressor power by bringing the condensing and evaporating temperatures closer together. From a design standpoint, compressor irreversibility can be reduced independently. Recent advances in the market have led to the use of scroll compressors. Replacing the reciprocating compressor used in this study by a scroll unit could increase the coefficient of performance.

Table 3
Exergetic calculation results provided for the DX-SAHP system

Item no.	Name of element	Rate of exergy destruction (irreversibility) $\dot{E}x_{\mathrm{dest}}(\dot{I})$ (kW)	Exergetic product rate $\dot{P}$ (kW)	Exergetic fuel rate $\dot{F}$ (kW)	Exergy efficiency ε (%)	Fuel depletion rate δ (%)	Relative irreversibility X (%)	Productivity lack ξ (%)	Exergetic factor f (%)
1	Compressor	1.18	1.66	2.85	58.48	14.27	57.06	23.08	34.38
2	Condenser	0.26	1.75	2.96	59.16	3.17	12.69	5.13	35.78
3	TEV	0.20	1.56	1.75	88.87	2.36	9.42	3.81	21.17
4	Solar collector	0.26	0.13	0.53	23.81	3.14	12.54	5.07	6.35
5	Circulating pump	0.17	0.02	0.19	10.74	2.07	8.29	3.35	2.32
	Overall system	2.07	5.12	8.28			100.00		100.00

Irreversibilities in the condenser occur due to the temperature differences between the heat exchanger fluids, pressure losses, flow imbalances and heat transfer with the environment. The irreversibility associated with the TEV is due to the pressure drop of the refrigerant passing through it. The only way to eliminate the throttling loss is to replace the capillary tube with an isentropic turbine (an isentropic expander) and to recover some shaft work from the pressure drop.

4. Conclusions

In this study, the studies on DX-SAHP systems were reviewed, while energy and exergy balance equations were derived for an illustrative example that is located in the Solar Energy Institute in Ege University, Izmir, Turkey.

The main conclusions, which may be drawn from the results of the present study, are listed below:

(a) Solar collectors used in studies were mostly bare flat-plate collectors. Very few of the studies used different types (e.g. glass covered, selective surface, PV/T) of the collectors. Bare flat-plate collectors were usually enough to generate the heat required to evaporate the refrigerant.
(b) Collector area used in the studies ranged from 1.82 to 10.5 m^2. The area of the collector used depends on the need of heat. Water-heating systems require less collector area, while space-heating and A/C systems need more due to more heat demand and thus increasing heat requirements.
(c) Traditional refrigerants, such as R-12 and R-22, were used in many of the studies conducted earlier. But as we approach to the near past, it can be seen that alternative refrigerants, namely R-134a, R-404a, R-407c and R-410a, were beginning to be used due to environmental regulations.
(d) The performance indicator of heat pumps, namely COP, ranged from 1.5 to 9. Taking into consideration that COP of traditional heat pumps can be assumed to be 2.3–3, higher values can be achieved by using direct expansion solar collectors as the evaporator in the heat pump cycle.
(e) Very few data were available about the collector performance in the studies. Existing data showed that collector efficiency ranged from 40% to 75%.
(f) Both theoretical and experimental analyses were performed in most of the studies.
(g) Energy analysis was performed in many of the studies, while exergy analysis can be seen in very few of them.
(h) Since space heating and A/C applications have more requirements, many of the systems were to be used in water heating applications.
(i) The theoretical calculations made on the illustrative example indicated that the exergy efficiency of the individual components of the DX-SAHP system ranged from 10.74% to 88.87%.

Acknowledgment

The authors would like to thank Ege University Research Fund for their financial support.

References

[1] Hulin H, Xinshi G, Yuehong S. Theoretical thermal performance analysis of two solar-assisted heat-pump systems. Int J Energy Res 1999;23:1–6.
[2] Freeman TL, Mitchell JW, Audit TE. Performance of combined solar-heat pump systems. Sol Energy 1979;22:125–35.
[3] Macarthur JW. Theoretical analysis of the dynamic interactions of vapor compression heat pump. Energy Convers Manage 1984;24:49–66.
[4] Duffie JA, Beckman WA. Solar engineering of thermal processes, 2nd ed. New York: Wiley, Inc.; 1991.
[5] Kaygusuz K, Comakli O, Ayhan T. Solar-assisted heat pump systems and energy storage. Sol Energy 1991;47:383–91.
[6] Kuang YH, Sumathy K, Wang RZ. Study on a direct-expansion solar-assisted heat pump water heating system. Int J Energy Res 2003;27:531–48.
[7] Ito S, Miura N, Wang K. Performance of a heat pump using direct expansion solar collectors. Sol Energy 1999;65:189–96.
[8] Charter WWS, Taylor LE. Some performance characteristics of a solar boosted heat pump. Proceedings of the I. I. R. Conference, Melbourne, Australia 1976:641–8.
[9] O'Dell MP, Mitchell JW, Beckman WA. Design method and performance of heat pumps with refrigerant-filled solar collectors. J Sol Energy Eng, Trans ASME 1984;106:159–64.
[10] Chaturvedi SK, Shen JY. Thermal performance of a direct expansion solar-assisted heat pump. Sol Energy 1984;33:155–62.
[11] Chaturvedi SK, Abazeri M. Transient simulation of a capacity-modulated, direct-expansion, solar-assisted heat pump. Sol Energy 1987;39:421–8.
[12] Chaturvedi SK, Chen DT, Kheireddine A. Thermal performance of a variable capacity direct expansion solar-assisted heat pump. Energy Convers Manage 1998;39:181–91.
[13] Torres-Reyes E, Picon Nunez M, Cervantes de Gortari J. Exergy analysis and optimization of a solar-assisted heat pump. Energy 1998;23:337–44.
[14] Torres-Reyes E, Cervantes de Gortari J. Optimal performance of an irreversible solar-assisted heat pump. Exergy 2001;1:107–11.
[15] Hawlader MNA, Chou SK, Ullah MZ. The performance of a solar-assisted heat pump water heating system. Appl Therm Eng 2001;21:1049–65.
[16] Cervantes JG, Torres-Reyes E. Experiments on a solar-assisted heat pump and an exergy analysis of the system. Appl Therm Eng 2002;22:1289–97.
[17] Chyng JP, Lee CP, Huang BJ. Performance analysis of a solar-assisted heat pump water heater. Sol Energy 2003;74:33–44.
[18] Gorozabel Chata FB, Chaturvedi SK, Almogbel A. Analysis of a direct expansion solar-assisted heat pump using different refrigerants. Energy Convers Manage 2005;46:2614–24.
[19] Ito S, Miura N, Takano y. Studies of heat pumps using direct expansion type solar collectors. ASME Trans 2005;127:60–4.
[20] Kuang YH, Wang RZ. Performance of a multi-functional direct-expansion solar-assisted heat pump system. Sol Energy 2006;80:795–803.
[21] Ozgener O, Hepbasli A. Experimental performance analysis of a solar-assisted ground-source heat pump greenhouse heating system. Energy Build 2005;37:101–10.
[22] Kotas TJ. The exergy method of thermal plant analysis. Essex, UK: Anchor Brendon Ltd.; 1985.
[23] Ozturk HH. Experimental determination of energy and exergy efficiency of the solar parabolic cooker. Sol Energy 2004;77:67–71.
[24] Singh N, Kaushik SC, Misra RD. Exergetic analysis of a solar thermal power system. Renew Energy 2000;19:135–43.
[25] Petela R. Exergy of undiluted thermal radiation. Sol Energy 2003;74:469–88.